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# ROTARY RADIAL PISTON MACHINE

## DESCRIPTION

5 The present invention relates to a radial piston type of rotary displacement machine.

While the complement of this description deals with a radial piston type of rotary displacement machine functioning as a pump  
 10 or a motor operated on a working fluid (e.g. air, water, oil), it should be understood that the teachings of this invention would equally apply to an internal combustion type of displacement machine, i.e. a rotary displacement machine where a combustible mixture is conventionally ignited within its radial cylindrical cham-  
 15 bers.

Radial piston rotary displacement machines have long been known which comprise:

- a supporting structure;
- 20 - a centrally mounted distributor;
- a rotating unit consisting of a rotor provided with a number of radially extending cylindrical chambers, wherein each chamber contains a respective piston mounted for sliding movement in a first direction along a first axis coaxial with the longitudinal center-  
 25 line of the respective cylindrical chamber;
- means of bucking the radial thrust from the pistons, said means forming a bearing in combination with an inner ring;
- support means carrying the rotating unit; and
- alignment means for maintaining the coaxial relationship of the  
 30 distributor to the rotor.

The following basic problems are encountered with such rotary volumetric machines of conventional design:

- (1) since the piston head is in spot contact with the inner sur-  
 35 face of the bearing, unacceptable concentrated loading is in-

curred, so that the design can only be adopted on machines having small-diameter pistons that are operated on relatively low pressures;

(2) the spot contact makes adequate hydraulic balancing impossible  
5 to achieve;

(3) no pressure surge control is provided for the piston; accordingly, any pressure drops in the hydraulic circuits are liable to result in the piston jumping off the bearing ring and producing knock that may harm the piston head as well as the thrust ring;

10 (4) a rotary displacement machine of this design has no arrangements for synchronizing the rotor and thrust ring rotations and preventing the piston heads from rubbing against the inner surface of the ring;

(5) the pistons of a machine of this design mount no seal rings;

15 (6) the rotor may come in frictional contact with the distributor, thereby lowering the overall mechanical effectiveness of the machine; and finally

(7) the distributor timing to the piston stroke cannot be adjusted.

20 A primary object of this invention is, therefore, to keep the piston under control without letting the piston lose contact with the surface of the thrust ring.

In addition, additional object of this invention is to provide a radial piston rotary displacement machine that has none of the  
25 drawbacks mentioned above.

This object is achieved by a radial piston rotary displacement machine according to Claim 1.

The invention will now be described with reference to the accompanying drawings, which show a non-limitative embodiment of the  
30 invention, in which:

- Figure 1 is a longitudinal cross-section taken through the radial piston rotary displacement machine of this invention;

- Figure 2 is a transverse cross-section taken along line A-A in  
35 Figure 1;

- Figure 3 shows a substantially cylindrical distributor incorporated in the rotary displacement machine of Figures 1 and 2;
- Figure 4 shows a piston incorporated in the rotary displacement machine of Figures 1 and 2;
- 5 - Figure 5 shows an engagement slide rail incorporated in the rotary displacement machine of Figures 1 and 2;
- Figure 6 shows the thrust ring (inner ring) of a rotor bearing incorporated in the rotary displacement machine of Figures 1 and 2; and
- 10 - Figure 7 shows a device synchronizing the rotation of the rotor and that of the bearing inner ring.

Note should be made that in the drawing figures, only such mechanical details as are necessary to an understanding of this invention are shown and referenced.

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Shown at 10 in Figures 1 and 2 is a radial piston rotary displacement machine according to the invention.

The machine 10 comprises a main body 11 that is configured into a substantially closed shell by a cover 12. The main body 11 and  
20 its cover 12 are held together by screw fasteners 13 and 14.

As shown in Figure 1, the bolt 13 (also useful to secure the machine 10 on a supporting structure, not shown) is passed here through clearance holes 11a and 12a formed through the main body 11 and the cover 12, respectively, and the screw 14 is threaded  
25 into two threaded holes 11b and 12b which are also formed in the body 11 and the cover 12. The embodiment shown has four bolts 13 (only one being shown in Figure 1) and two screws 14 (only one being shown in Figure 1).

The space between the main body 11 and the cover 12 accommodates a distributor 15 of whatever fluid. The distributor 15 is  
30 substantially cylindrical in shape about an axis A, and is illustrated in greater detail in Figure 3.

As explained hereinafter, the distributor 15 is mounted to float within the space defined by the cover 12, but is not rotated  
35 about the axis A that also forms its longitudinal centerline.

Furthermore, the distributor 15 is encircled by a rotating unit 16 (Figure 1) which comprises a rotor 17 arranged to turn about the same axis A as the distributor 15.

The rotor 17 is formed conventionally with a plurality of radially extending cylindrical chambers 18 (only two being shown in Figure 1), each chamber being adapted to receive a respective piston 19 for movement along a radial direction (a) as shall be subsequently better illustrated .

As shown in Figures 1 and 3, the distributor 15 is formed with two slots 15a, 15b and four cutouts 15c-15f. The cutout pairs 15c, 15f and 15d, 15e are each provided with a bracing rib 20 and 21.

As can be seen from the combined Figures 3a, 3b and 3c, the slot 15a is communicated to the cutouts 15d, 15e by a pair of conduits 22 and 23, the fluid connection between the slot 15b and the cutouts 15c, 15f being established by conduits 24 and 25.

The conduits 22-25 open at their left end as shown in Figure 3a.

As depicted in Figures 1 and 2, each radial cylindrical chamber 18 will be placed sequentially in fluid communication with the cutouts 15c-15f as the rotor 17 turns about the axis A.

In the embodiment shown, assuming the machine 10 is to be operated as a hydraulic motor, the machine 10 would be supplied pressurized oil through the conduits 22, 23, the oil being then discharged through the conduits 24, 25. For the purpose, the cover 12 is provided with an oil intake device 26 effective to deliver the pressurized oil incoming from a remote source, and with an oil discharge device 27.

In particular, the intake device 26 comprises the aforementioned cutout 15a in the distributor 15 (Figures 3a-b), a corresponding groove 26a formed in the cover 12 at an offset location from the axis A, and an intake port 26b.

Likewise, the discharge device 27 comprises the aforementioned cutout 15b in the distributor 15 (Figures 3a-b), a corresponding groove 27a formed in the cover 12 at an offset location from the

axis A, and a discharge port 27b.

In this example, the oil inflow runs in the direction of arrow F1, and the oil outflow in that of arrow F2.

5 As shown in Figure 1, each piston 19 is engaged with the thrust ring 28 of a bearing 29 by means to be described.

The ring 28 is, moreover, an integral part of the rotating unit 16, which unit includes, as said before, the rotor 17 and pistons 19.

10 In other words, the thrust ring 28 also forms the inner ring of an integral bearing 29 that additionally comprises an outer ring 30 and two sets of cylindrical rollers 31 conventionally disposed between the inner ring 28 and the outer ring 30.

The combination of the multiple rollers 31 and outer ring 30  
15 provides a means of bucking the radial thrust forces from the pistons 19.

Also, integral bearing means C1, C4 are arranged to support the rotating unit 16 and take up the forces from the pistons 19, and integral means of alignment C2, C3 are arranged to maintain the  
20 coaxial relationship of the distributor 15 and rotor 17 along the axis A, this alignment being made crucial by the provision of an odd number of pistons 19.

The term "integral bearing" encompasses here a design where the bearing races are formed directly on the members of the machine  
25 10, i.e. no intermediate rings are provided.

Advantageously, the bearings C1-C4 are an interference fit to prevent creeping of the axis A of distributor 15.

The outer ring 30 is held stationary and has a centerline B (Figure 1) generally offset from the axis A; it can be shifted radially by means of an adjuster 33 (Figure 2) intended for adjusting  
30 the offset EC (Figure 1) between the lines A and B.

The adjuster 33 is a conventional design and no further described herein. In addition, the adjuster 33 may be a mechanical, hydraulic, electromechanical, or otherwise operated device.

35 The rotating unit 16 is driven conventionally. In an applica-

tion where the machine 10 is operated in the hydraulic motor mode, head and delivery rate are converted within the machine 10 to rotary power by the rotating unit 16, specifically the rotor 17, due to the piston heads 19 urging against the ring 28, and due to the thrust forces being offset by the amount EC. This offset EC is essential to the rotation of the unit 16. Should the offset EC be nil, no rotation would be possible because the thrust ring 28 would enter a stalled condition.

10 As mentioned before and shown in Figure 4, each piston 19 is shaped for engagement with the ring 28. Sliding engagement is achieved by contour shape, comprising a slide rail 43 (Figure 5) attached to the rotating ring 28 by a screw 44. A slide 45 (Figure 4) is formed integrally on the head of the piston 19 to allow  
15 small movements of the piston 19 relative to the ring 28. As shown in Figure 2, the movements of the slide 45 along the slide rail 43 take place in a straight direction along an axis (b) perpendicular to the aforesaid axis (a) along which the piston 19 moves radially. The axis (a) also is, as mentioned, the centerline of the radial cylindrical chamber 18 in which the piston 19 is movable.  
20

In other words, the slide rail 43 extends perpendicularly to the direction of the axis (a), and ensures that no cocking of the axis (a) of the piston 19 may occur with respect to the axis of the chamber 18.

25 These movements of the piston 19 along the axis (b) are needed to adapt the piston setting for the geometrical conditions that prevail during the rotation of the rotating unit 16. The slide rail 43 of this embodiment is illustrated in greater detail in Figure 5.

30 The slide rail 43 comprises a body 43a which is formed with a threaded hole 43b receiving the screw 44 threadably therein (Figure 1). Two jaws 43c jut out of the body 43a to engage the slide 45, the latter being as mentioned integral with the piston 19.

In an embodiment not shown, the slide rail 43 is integral with  
35 the ring 28.

The function of the slide rail 43 made integral with the ring 28, and of the slide 45 that is formed integrally with the piston head 19, is fundamental to this invention. As previously mentioned, in one of the commercially available embodiments, the head 5 of the piston 19 is mounted to merely rest onto the thrust ring 28. Thus, surges involving a pressure drop through the hydraulic circuit are liable to cause the piston 19 to move away from the surface of the ring 28. As the rotational movement goes on, the piston 19 is bound to meet geometrical and kinematic conditions 10 that will urge it back against the inner surface of the ring 28, thereby initiating a series of piston 19 knocks on the ring which may seriously harm the piston head 19 and the inner ring 28 surface as well.

Accordingly, it matters in this invention that the head of the 15 piston 19 cannot become detached from the inner surface of the ring 28, so that pressure surges through the hydraulic circuit will not harm the above parts.

Also, the inner ring 28 may advantageously be provided a substantially sinusoidal shape, such that the two sets of rollers 31 20 can be received in two side races, with the roller sets located on either side of the slide rail 43.

Referring back to Figure 4, it can be seen that the piston 19 and its attached slide 45, is formed with a pair of lightening 25 holes 46 drilled crosswise through it for reduced inertia. In addition, the piston 19 is drilled along the axis (a) with a small hole 47 allowing a determined amount of oil to flow into a recess 48 in the head of the piston 19 itself. The amount of oil admitted through the hole 47 is to balance out hydraulically the forces 30 acting on the piston 19.

As shown in Figure 4b, the centerlines of the holes 46 extend parallel to each other crosswise to the axis (a) of the hole 47. This allows the piston 19 to be lightened at no consequence for the diameter of the hole 47. In another embodiment not shown, the 35 holes 46 do not go through, but converge radially on the hole 47

to a point somewhat short of it.

The outer surface of the piston 19 is formed with a groove 49 (Figures 4a-b) that can receive a seal ring (not shown). In addition, two cutouts 49a are formed opposite to each other at the location of the groove 49, as shown in Figures 4a-c. These cutouts 49a enable said seal ring (not shown) to be installed.

As shown in Figures 4a-b, the far surface from where the recess 48 is shaped to restrict the clearance between the skirt of the piston 19 and its chamber 18.

Figure 4e shows an alternative embodiment of the piston 19 that differs from that shown in Figures 4a-d only by the configuration of one of the front faces of the piston 19.

In this embodiment, the recess 48 shown in Figures 4a-b is replaced by a groove 49b that matches the contour of the head surface of the piston 19. This groove 49b is in fluid communication with the hole 47 through two radial canalizations 49c. This configuration affords increased surface area for improved hydrodynamic effect where this is required.

A modified embodiment of the ring 28 is shown in Figure 6, wherein the ring 28 is split to provide two separate portions 28a, 28b that can be joined together by means of a set of screws 28c (only two screws 28c being shown in Figure 6).

This embodiment allows the rotor 17 to be inserted into the portion 28a complete with pistons 19 and associated slides 45, without incurring interference with the small diameter of the portion 28a. This allows the system displacement to be increased substantially, since longer cylinders 19 and longer strokes can be used.

An outer ring 30 formed of two parts that can be assembled together conventionally, e.g. by welding along their centerline, could be provided instead.

As shown in Figure 1, moreover, the piston 19 is quite short, and part of the engaging arrangement to the inner ring 28, with the piston 19 at either dead center (top half of Figure 1), is



nested within the respective chamber 18. This greatly reduces the machine 10 cross-section outline, and with it the inertia of the moving masses during rotation of the rotating unit 16.

Figure 1 shows that the rotor 17 carries the distributor 15  
5 through the bearing pair C2, C3.

Furthermore, as any of the bearings C1-C4 and bearing 29, disk-cage bearings GAB may be used to advantage, as described in WO 01/29439 and only shown here as to bearing 29. Optionally, the cages GAB may be closed, viz. unsplit, cages rather than split  
10 cages as described in the above document.

By using unsplit disk cages GAB for the bearings of the machine 10, the life span of the latter can be extended considerably. The unsplit disk cage GAB is effective to bring the loss of rollers down to 7-10%, as against 30% with conventional cage designs. This  
15 represents an important improvement in terms of allowable loading and speed, and consequently of output power. Although each cage GAB is shown mounted centrally of its associated set of rollers 31, different arrangements may provide for the cage GAB to be mounted peripherally of the roller set 31.

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In the embodiment shown, the spacing of these bearings C2 and C3 along the axis A is quite small. Accordingly, deflection of the distributor 15 to rub against the rotor 17 is effectively avoided, even where the clearance between these parts is quite narrow.

25 As shown in Figures 1 and 3, the surface of the distributor 15 included between the two bearings C2 and C3 and involved in the fluid distribution process has portions S1', S3', S1", S3" facing the cutouts 15d, 15e and cutouts 15c, 15f, respectively.

These portions S1', S3', S1", S3", and the corresponding sur-  
30 faces s2 and s4 of the recess CAV in the rotor 17 (Figure 1) may be conical rather than cylindrical in shape as shown in the drawings. Clearly S1' and S3' have a single cone generatrix line, as have the pair S1', S3' on one side, and the pair S2, S4 on the recess CAV side. In this way, the amount of oil that is allowed to  
35 leak into the distribution area can be adjusted by shifting the

distributor 15 along the axis A. Consequently, a virtually complete seal-off could be provided instead.

Alternatively, compromise arrangements could be provided, e.g. one that would admit significant leakage of pressurized oil in order to lubricate other system parts.

The oil pressurization at the cutouts 15d, 15e is bound to generate radial loads that would be transferred to some extent onto the surfaces S1" and S3" of the distributor 15. Likewise, pressurization of the oil at the cutouts 15c, 15f is bound to generate radial loads that would be transferred to some extent onto the surfaces S1' and S3' of the distributor 15. This makes counterbalancing such radial loads hydraulically a necessity if rubbing contact of the distributor 15 against the recess CAV in the rotor 17 is to be prevented. For the purpose, and as shown in Figures 3a and 3c, canalizations are provided such as the canalization CAN1 that place the conduit 25 in fluid communication with the surface S3' of the distributor 15. The surfaces S1', S2" and S3" are similarly communicated to their respective conduits. For example, the surface S3" is placed in fluid communication with the conduit 22 through a canalization CAN2 (Figure 3c). In this way, a passage is created for the fluid between the surfaces S1', S3', S1", S3" on the one side, and the surfaces S2, S4 of the recess CAV, on the other.

This passage is useful to balance out the hydraulic forces.

As a result, the bearings C2 and C3 are only called upon to bear the alternating loads from the interconnection area between the distributor 15 and the radial cylindrical chambers 18, in addition to loads due to any imprecise balancing.

Also, this arrangement is innovative in that the distributor portion 15 found to the left of the bearing C2 is free to float under the cover 12. A hole F in the cover 12 accounts for the floating feature of the distributor 15.

To prevent oil from leaking through a clearance between the outer surface of the distributor 15 and the surface of the hole F,

ring seals AN are provided at either ends of the devices 26, 27. These ring seals AN fit in closed seats formed in the surface of the hole F in the cover 12. "Closed seat" refers here to an annular groove formed in the cover 12. Advantageously, moreover, the  
5 rings AN are made of appropriate materials (steel, Teflon(r), etc.) for the pressure, temperature, and amount of clearance anticipated.

The floating feature of the distributor 15 is also essential to this invention.

10 In fact, the outer surface of the distributor 15 must be prevented from contacting the inner surface of the rotor 17 at all cost. By inhibiting all contact, no frictional drag would be incurred, and the efficiency is maximized.

By thus preventing all contact, the contamination problem due  
15 to various particles being introduced with the oil is also solved.

All the moving parts of this invention are, advantageously but not necessarily, case hardened parts to a hardness of about 60 HRC. However, the distribution surfaces S1', S1", S3', S3", S2 and S4 adjacent to the cutouts 15c-f (see also Figure 3c) should ad-  
20 vantageously have hardness of 1400 HV or above.

By providing the bearings C2, C3 and the balanced hydraulics as described hereinabove, any use of anti-friction metals such as bronze and other copper alloys, cast iron, aluminum alloys, etc.  
25 in the construction of the rotor 17, for example, is made unnecessary.

By providing a floating distributor 15, the machine 10 can be timed for optimum performance.

Any piston machine presents the problem of variable timing. The  
30 chamber injecting or discharging functions require to be advanced or retarded relative to the dead centers according to such factors as pressure, rotation, etc..

By having the distributor 15 unconnected to any other parts, it can be turned through a given angle using means not shown, to ad-  
35 vance or retard the intake and discharge phases as required.

Phase adjustment may be made necessary by the presence of clearance, and by a varying pressure, rotation, displacement, etc.. As the intake and discharge phases are optimized, the system will run quieter and vibration become trivial. In addition, the  
5 bearings extend their life span, and the output torque of the machine 10 is made steadier.

Any resetting of the distributor 15 would be a trial-and-error process, because each machine 10 is to be timed separately.

Also, the motion of the rotor 17 is reversed when the distributor  
10 15 is rotated 180 degrees.

In addition to the above angle adjustment, and if machine 10 is operated in the pump mode as well as the motor mode, so that the distributor 15 is to function in either situation, axial adjustment (along axis A) must be performed using two grooves GF offset  
15 from the centerline M (see Figure 3a).

Thus, for quiet vibration-less running, two grooves GF should be provided for use, the one when the machine 10 is operated in the pump mode and the other when in the motor mode.

Position shifting along the axis A for selection of the groove  
20 GF is also significant when the machine 10 is operated as a clockwise or counterclockwise rotating pump.

A person skilled in the art will recognize that by enabling the distributor 15 to be shifted both angularly and axially along axis A, a variety of demands on the machine 10 can be filled.

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Also, the invention includes a cross coupling 50 (Figures 1 and 7), whereby the ring 28 of the bearing 29 against which the pistons 19 are urged can turn in perfect synchronization with the rotor 17.

30 The cross coupling 50 also effectively minimizes the requirements of the piston 19 for guide inside its chamber 18.

"Guide" is used here to indicate that portion of the chamber wall which remains in contact with the piston surface when the piston 19 is moved to its farthest position out of the chamber 18.

35 The cross coupling 50 and the slides 45 keep the piston 19

aligned to the chamber 18, so that short guides can be used and radial bulk reduced.

By contrast, in state-of-art embodiments having no cross coupling 50, a piston guide whose length amounts to 50% and 100% of the piston 19 diameter must be provided.

More particularly, the cross coupling 50 comprises, as shown best in Figure 7, a plate 50a advantageously made of treated steel. The plate 50a is formed with a center hole 50b, and two peripheral notches 50c receiving two cogs 52 (Figure 1) of the ring 28. Two prismatic guides 50d are arranged to guide the movements of two cogs 53 (only one being shown in dash lines in Figure 1) integral with the rotor 17. The prismatic guides 50d are connected to the substantially rectangular center hole 50b. The shape of the center hole 50b is effective to only allow movement of the cogs 53 along the direction of the long side of the center hole 50b.

It will be appreciated that other conventional devices, such as a constant velocity joint, gear pairs, etc. could be employed to keep the ring 28 synchronized with the rotor 17.

Finally, in the tight fit of the distributor 15 and rotor 17, the rotor mating surface may advantageously be nitrided to have it withstand local heating and obviate seizure.

Lastly, the rotary displacement machine described above could have the roll bearings 29 or C1 or C4 replaced with plain bearings having a sliding means formed of at least one layer of an anti-friction plastics material bonded through an additional layer of a porous metal, on one of the contacting parts or an intervening metal element.

The advantages of this rotary displacement machine 10 are:

- compared with current displacement machines, approximately 70% less friction; the range of displacement machines that can be produced is therefore extended from 1 cm<sup>3</sup> capacity to more than 30,000 cm<sup>3</sup>, while retaining a high efficiency;
- for the same size, this system affords a higher power output

- than conventional machines, since it can attain higher speeds;
- both the working pressure and the power output can be increased by virtue of a lower specific loading, particulate contaminants would cause no significant harm since all the moving parts are
  - 5 surface hardened;
  - the thrust ring and rotor rotations are exactly synchronized, leaving the pistons and engagement arrangements unharmed;
  - a distributor which is mounted floating;
  - the machine timing can be adjusted by rotating and/or shifting
  - 10 the distributor axially;
  - the rotary displacement machine performs equally well in the pump and motor modes;
  - when the rotary displacement machine is operated in the pump mode, the pump may be made to turn clockwise or counterclockwise
  - 15 by merely changing the axial placement of the distributor.

While the machine of this invention has been described essentially as a hydraulic motor or a hydraulic pump, it should be understood that the machine could also function as a hydraulically

20 operated speed variator.

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## CLAIMS

1. A rotary displacement machine (10) with radial pistons (19); rotary displacement machine (10), comprising:
- 5 - a supporting structure (11, 12);
- a centrally mounted distributor (15);
- a rotating unit (16) consisting of a rotor (17) provided with a number of radially extending cylindrical chambers (18), wherein each chamber (18) contains a respective piston (19) mounted for
- 10 sliding movement in a first direction along a first axis (a) coaxial with the longitudinal centerline of the respective cylindrical chamber (18); and
- means (30, 31) of bucking the radial thrust from the pistons (19), said means (30, 31) forming a bearing (29) in combination
- 15 with a thrust ring (28);
- the rotary displacement machine (10) being characterized in that:
- said bearing (29) comprises a rotating inner ring (28), a stationary outer ring (30), and intervening rolling means (31), said
- 20 rotating inner ring (28) including engagement means (43, 45) for each piston (19), said engagement means (43, 45) allowing movement in a straight line along a first direction defined by a second axis (b) perpendicular to said first axis (a).
- 25 2. A rotary displacement machine (10) as claimed in Claim 1, wherein said engagement means (43, 45) are sliding engagement means.
3. A rotary displacement machine (10) as claimed in Claim 2,
- 30 wherein said engagement means (43, 45) comprise a slide rail (43) attached to said ring (28), and a slide (45) attached to the head of said piston (19), said slide (45) being a flat slide (45), so that the relative paths of movement of said slide (45) and said slide rail (43) are straight paths of movement along said axis
- 35 (b).

4. A rotary displacement machine (10) as claimed in any of the preceding claims, wherein the force of the piston (19) is transferred to the thrust ring (28) through a hydraulically balanced end surface.

5. A rotary displacement machine (10) as claimed in any of the preceding claims, wherein at least one of said pistons (19) is provided with a closed seal ring.

6. A rotary displacement machine (10) as claimed in any of the preceding claims, wherein at least one of said pistons (19) is facing said distributor (15) with a face shaped to fill unwanted clearance.

7. A rotary displacement machine (10) as claimed in any of the preceding claims, wherein at least one piston (19) is formed with at least one lightening hole (46).

8. A rotary displacement machine (10) as claimed in Claim 7, wherein the longitudinal axis of said hole (46) extends transverse to the axis (a) of the piston (19) and does not cross a hydraulic balancing hole (47) formed in the piston (19).

9. A rotary displacement machine (10) as claimed in any of Claims 3-8, wherein one of said pistons (19) locates fully inside the respective radial cylindrical chamber (18), and at least a portion of said slide rail (43) locates inside said radial cylindrical chamber (18).

10. A rotary displacement machine (10) as claimed in Claim 1, wherein at least one of said bearings (29, C1-C4) is an integral bearing.

11. A rotary displacement machine (10) as claimed in Claims 3



and 10, wherein the ring (28) has advantageously a sinusoidal shape, such that it can accommodate two sets of rolling bodies (31) in two side races, they being placed on one side of said slide rail (43).

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12. A rotary displacement machine (10) as claimed in Claim 10, wherein at least one of said bearings (29, C1-C4) mounts an unsplit disk cage (GAB).

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13. A rotary displacement machine (10) as claimed in Claim 12, wherein each unsplit disk cage (GAB) is mounted peripherally of the respective set of rolling bodies (31).

14. A rotary displacement machine (10) as claimed in Claim 10, wherein at least one of said bearings (29, C1-C4) mounts a plurality of rolling bodies in interference fit relationship.

15. A rotary displacement machine (10) as claimed in Claim 1, wherein said rotor (17) and thrust ring (28) are controlled to rotate synchronously by a synchronization device (50).

16. A rotary displacement machine (10) as claimed in Claim 15, wherein said synchronization device (50) is a cross coupling (50).

17. A rotary displacement machine (10) as claimed in Claim 1, wherein said distributor (15) is mounted floating in the portion carrying the cover (12).

18. A rotary displacement machine (10) as claimed in Claim 17, wherein the placement of said distributor (15) can be adjusted both angularly and axially along a longitudinal centerline (A).

19. A rotary displacement machine (10) as claimed in any of Claims 17 and 18, wherein at least a surface portion of a recess (CAV) provided on the rotor (17) has a conical shape allowing said

surface portions to fit together in different ways.

20. A rotary displacement machine (10) as claimed in Claim 17, wherein seal rings (AN) of metal are arranged to stop oil from  
5 leaking through the clearance gap between the outer surface of the distributor (15) and the surface of said hole (F) in said cover (12).

21. A rotary displacement machine (10) as claimed in Claim 20,  
10 wherein said rings (AN) are received each in a respective annular seat formed in the surface of said hole (F).

22. A rotary displacement machine (10) as claimed in Claim 1, wherein said cover (12) carries an intake device (26) and a discharge device (27), said intake and discharge devices (26, 27) being each formed with a respective offset groove (26a, 27a) from a centerline (A) of the distributor (15).

23. A rotary displacement machine (10) with radial pistons  
20 (19); rotary displacement machine (10), comprising:  
- a supporting structure (11, 12);  
- a centrally mounted distributor (15);  
- a rotating unit (16) consisting of a rotor (17) provided with a number of radially extending cylindrical chambers (18), wherein  
25 each chamber (18) contains a respective piston (19) mounted for sliding movement in a first direction along a first axis (a) coaxial with the longitudinal centerline of the respective cylindrical chamber (18); and  
- means (30, 31) of bucking the radial thrust from the pistons  
30 (19), said means forming a bearing (29) in combination with a thrust ring (28);  
- the rotary displacement machine (10) being characterized in that: said distributor (15) is mounted floating in the cover-carrying portion (12).

24. A rotary displacement machine (10) as claimed in any one of Claims 1 to 22, wherein at least one of the bearings (29, C1 or C4) for the rotor (17) and/or for coupling the inner and outer rings (28, 30) together provides frictional drag in which sliding  
5 means are provided which comprise at least one layer of an anti-friction plastics material bonded, through an additional layer of a porous metal, to one of the contacting parts or another intervening metal element.

10 25. A rotary displacement machine (10) as claimed in any of the preceding claims, wherein said rotor (17) has a nitrided surface in the area of coupling to said distributor (15).

26. A hydraulically operated speed variator, characterized in  
15 that it incorporates at least one machine (10) as claimed in any of the preceding claims.

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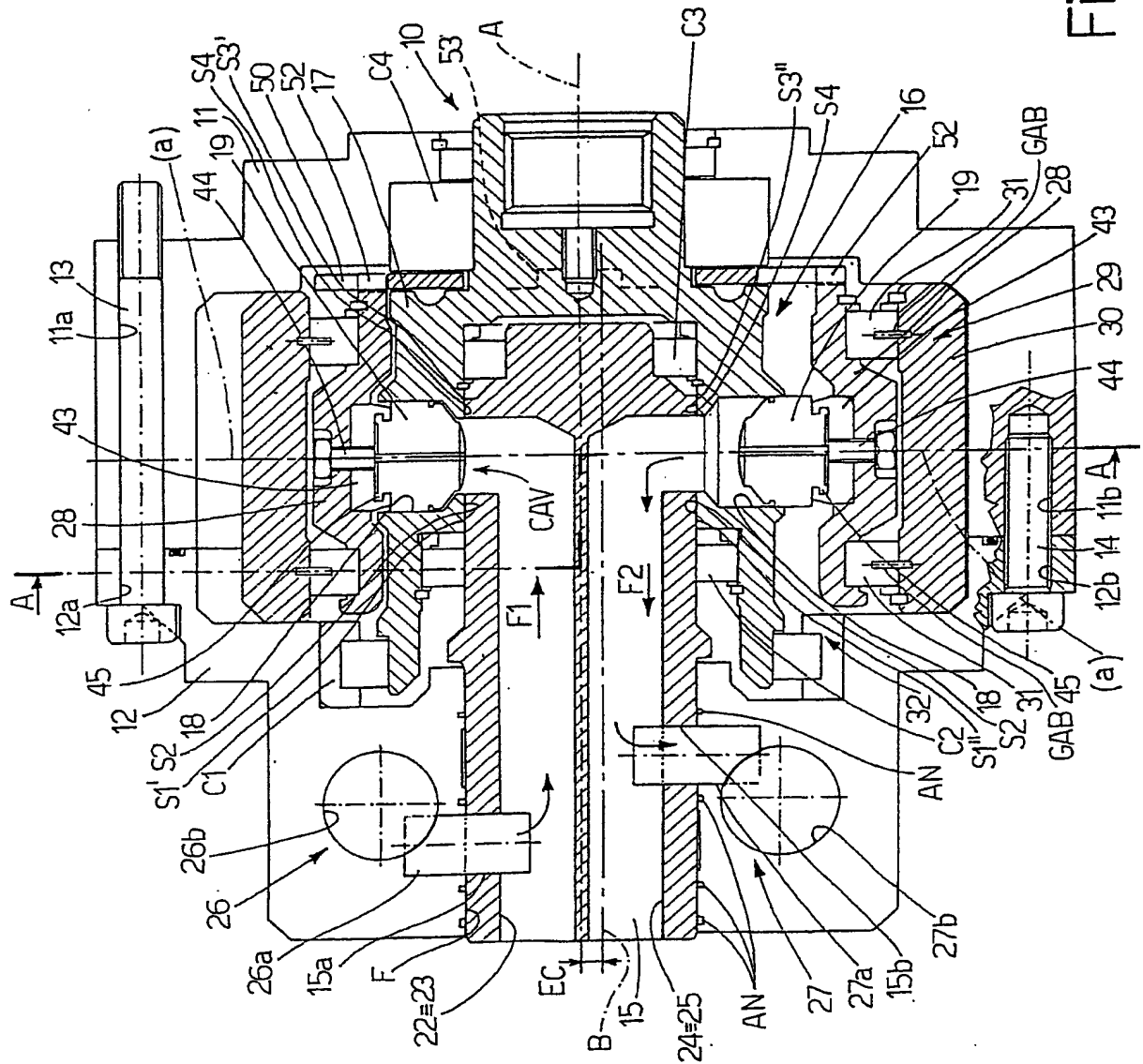
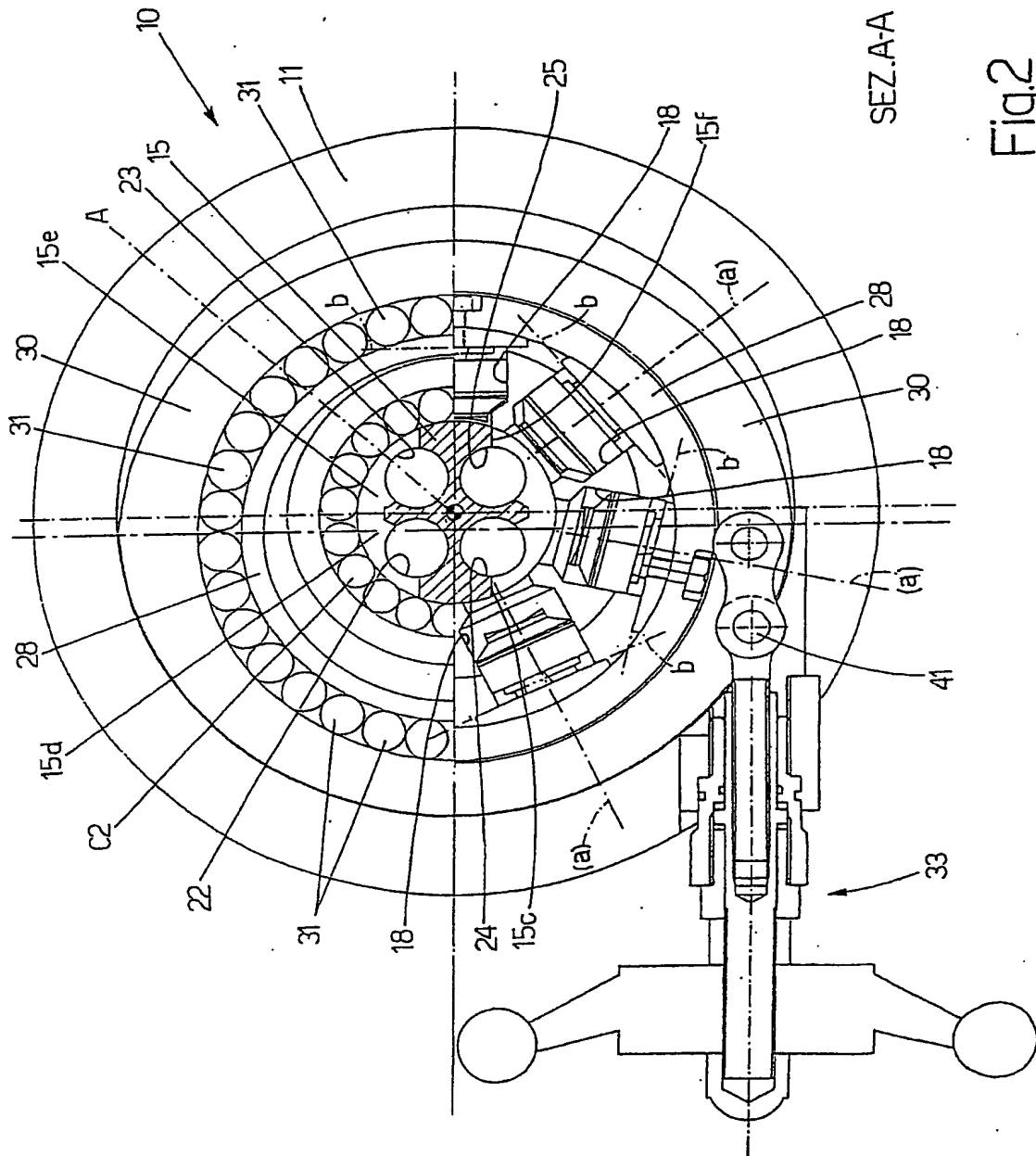


Fig.1



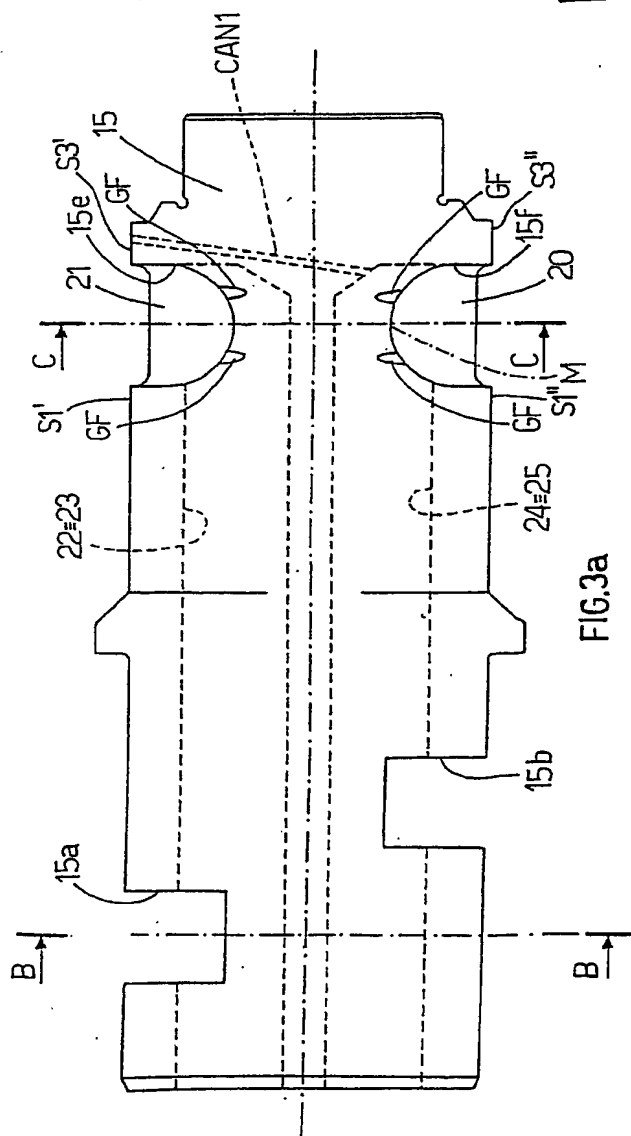
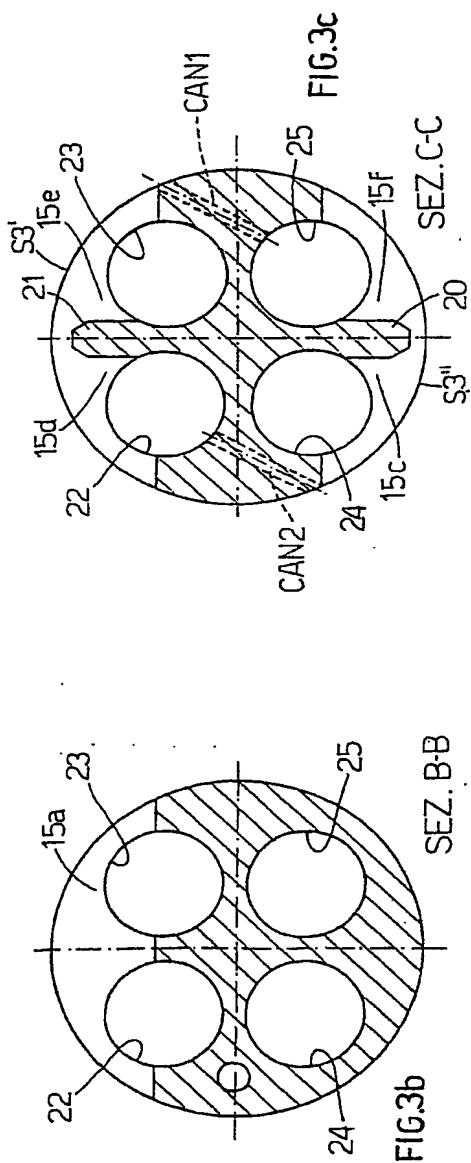


Fig.3

FIG.3a

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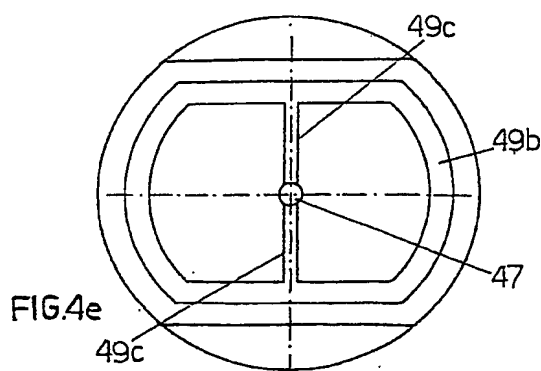
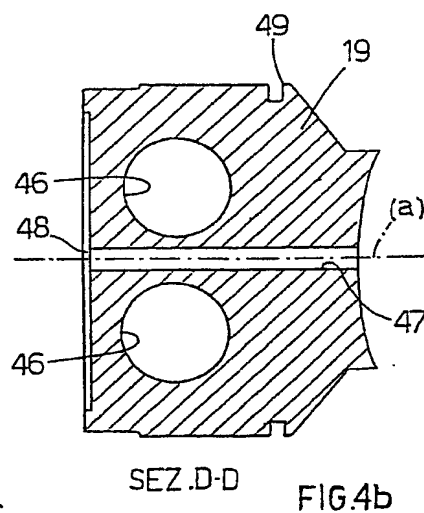
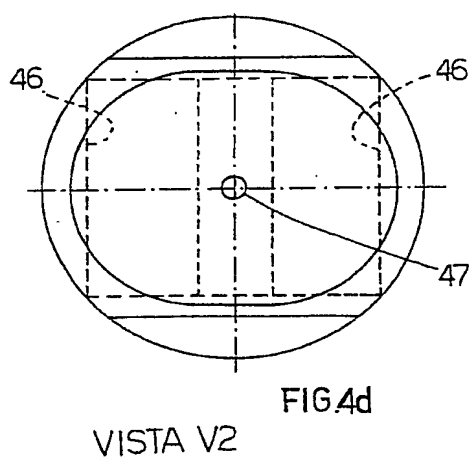
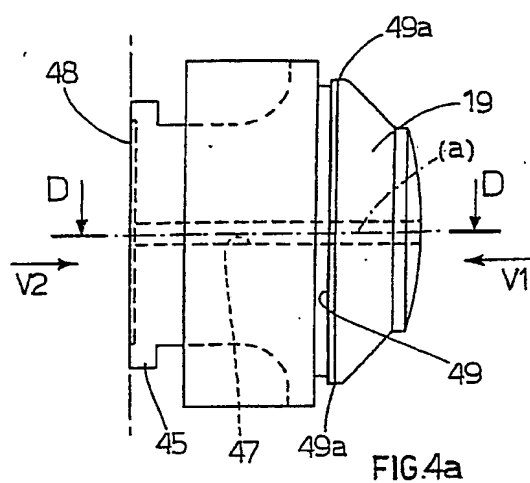
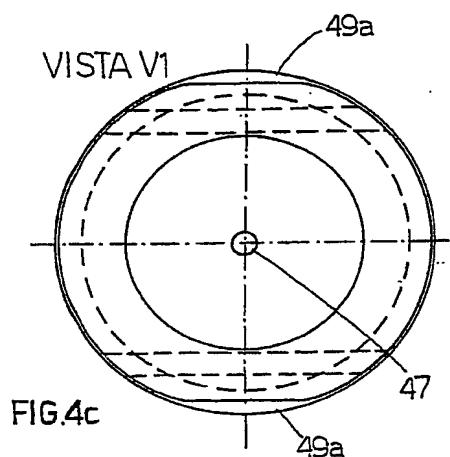


Fig. 4

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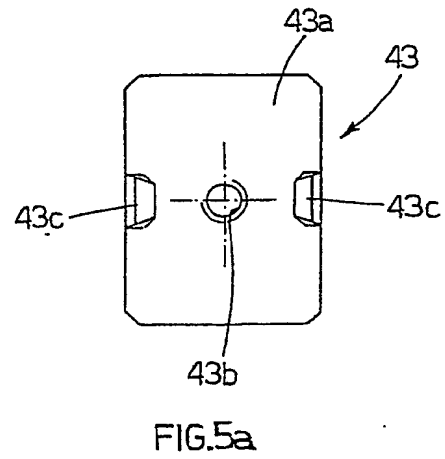
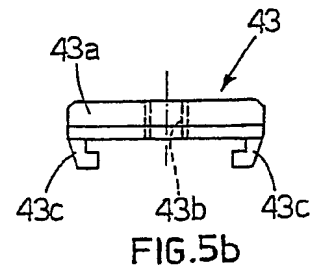
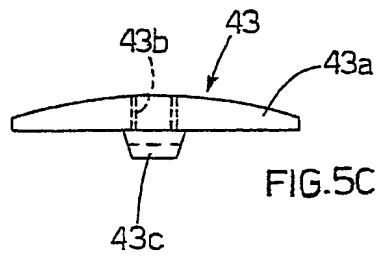


Fig.5



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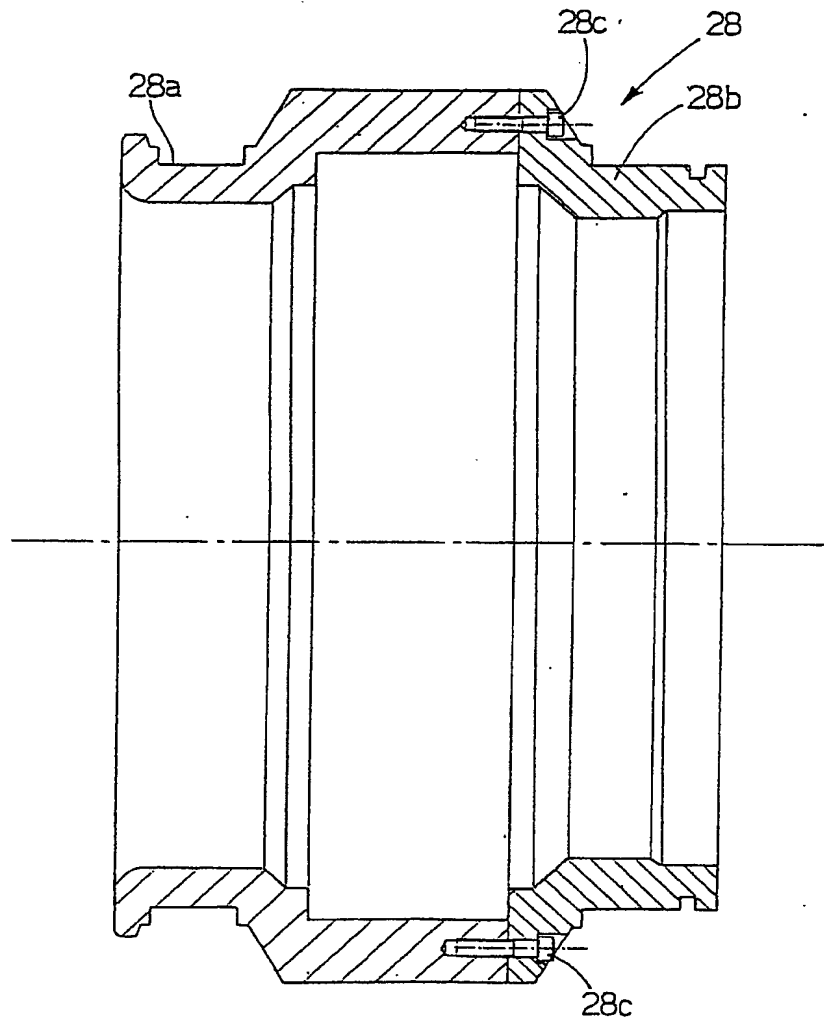


Fig.6

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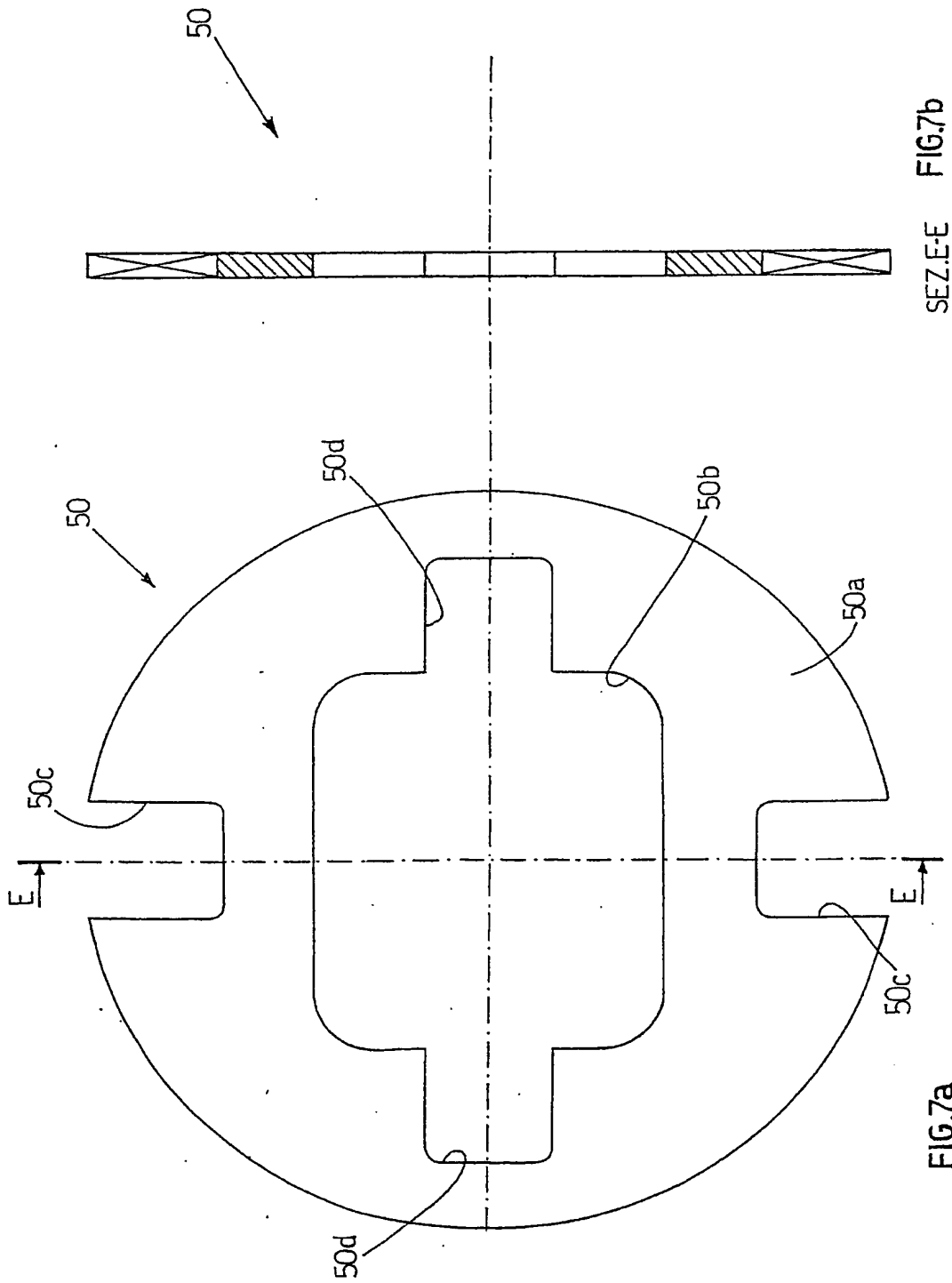


Fig.7